HIGH FREQUENCY FATIGUE FAILURE IN SILENCER/PULSATION DAMPERS FOR OIL-FREE SCREW COMPRESSORS

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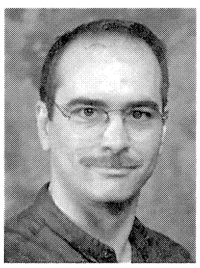

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ABSTRACT

Oil-free screw (OFS) compressors are capable of producing high-pressure pulsations in the discharge gas stream, over a frequency range of 200 to 3000 Hz. The fundamental pressure pulsation is due to the opening cycle of the compressor lobes at the discharge porting and is referred to as the pocket passing frequency (PPF). Fatigue failures have been observed in discharge silencers/pulsation dampers as well as downstream piping components in oil-free screw compressor service where the frequencies were greater than 1000 Hz. A careful review of the pulsation damper design has shown that attenuation effectiveness is limited to frequencies less than 1000 Hz. The principal purpose of the pulsation damper is to reduce the line pressure pulsations in order to lessen the chances of downstream piping failures. The pulsation damper does very little to attenuate high frequency noise levels that are transmitted to the area surrounding the compressor.

The principal purpose of this paper is to document failures of silencer/pulsation dampers in OFS compressor service and provide a root cause analysis of these failure modes. Pulsation dampers are not used at the inlet flange in many applications, due to the low-pressure pulsation in the inlet plenum of the compressor. It is not the intent of this paper to present any theoretical background in the design of acoustic dampers, but to provide an overview of the acoustic phenomenon and how it is used in screw compressor applications. The end result of this work will be to provide practical guidelines for specifying silencer/pulsation dampers for OFS compressor applications.

INTRODUCTION

Pulsation damper (PD) design consists of a carefully selected collection of passive attenuation components that decrease the pulsation amplitude of a discrete set of frequencies that are produced by the compressor. Pulsation dampers designed for API service are designed for an acoustic cutoff of approximately 1000 Hz. The pulsation pressure magnitude of frequencies above 1000 Hz is generally considered to be insufficient to cause piping damage. Measurements of pressure pulsation and piping vibration have shown that the higher order multiples over 1000 Hz can excite mechanical vibration, resulting in severe damage. Attenuation of high order multiples results in lower noise levels in the piping downstream of the silencer and reduces the source of excitation that can produce mechanical resonance.

COMPRESSOR PULSATION SOURCE THREAD OPENING SEQUENCE

Gas compression in the screw compressor takes place in three chambers within the compressor. One is the inlet plenum, which is the volume from the inlet flange to the inlet end-wall of the rotors. The second is thread volume along the rotors, which is bounded by the casing bore walls and the rotor end-planes. The third is the discharge plenum that extends from the porting at the discharge end-wall to the discharge flange. The pressure distribution during the discharge thread opening is the primary source of pressure pulsation at the discharge of the screw compressor. The volume between the threads of each rotor is filled in the inlet plenum. The threads are exposed at the inlet end-wall after the same volume is closed at the discharge end-wall. Gas from the inlet plenum starts to fill the threads as they rotate toward the convolution of the threads at the inlet port cusp (Figure 1). The inrush of gas into the threads produces a pressure reduction in the inlet plenum. This pressure change is one source of pressure pulsation at the inlet flange. The magnitude of the inlet pressure pulsation is much lower than that at the discharge port. The threads complete the inlet closing cycle when the pitch line of each rotor meets at the inlet-porting cusp (Figure 1). It is important to note that the discharge end-wall closes the open area on the discharge end of the lobes that are being filled.

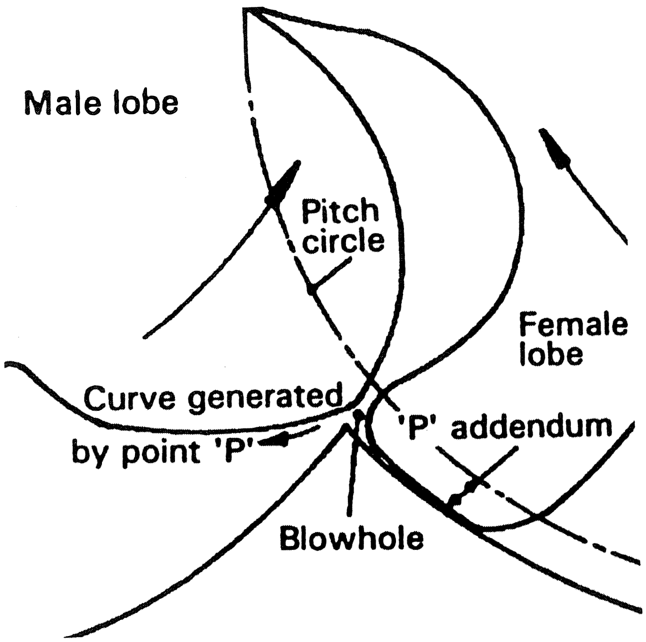

Figure 1. Rotor Profile Convolution at the Inlet Porting.

The compression of the gas takes place as the threads continue to close, thus reducing the enclosed volume. The shape of the discharge porting determines the pressure rise within the enclosed threads. Pressure in the discharge plenum begins to rise as the threads open at the discharge port. The pressure gradient produced by this thread opening sequence determines the magnitude of the primary pressure pulsation. The number of thread opening cycles per second determines the primary pulsation frequency, which is known as the pocket passing frequency (PPF). A screw compressor with a 4×6-rotor configuration (four male and six female lobes) that is driven by the male rotor at 3778 rpm has a fundamental PPF of 251 Hz.

Internal Sources of Resonance

There are several sources of pulsation excitation within the compressor pulsation damper system that generate frequencies above 1000 Hz. The exposed volume of gas within the rotor threads changes as a function of time at the inlet and discharge plenum. This change in volume results in a change in impedance at the inlet and discharge. The thread volume that is exposed to the downstream plenum causes an impedance mismatch that can, in itself, be a source of higher order harmonics of the fundamental. Changes in impedance occur at changes in the cross-section along the flow path of an acoustic wave. The opening at the end of the inlet tube to the PD produces a very large change in impedance along the process flow path. The discharge plenum and inlet side of the PD must be considered as a coupled acoustic system for evaluation of the resonant frequencies that could occur at the compressor discharge.

The amplitude of the pure harmonics decays exponentially as the frequency increases. Measurements taken during operation of a screw compressor in operation indicate that the frequency content of the pressure pulsation at the discharge includes high order multiples of the PPF up to 2750 Hz, approximately 11 times the PPF (Figure 2).

The pulsation measurements (shown in Figure 2) were taken on a compressor operating on an open loop with a straight section of pipe in place of the discharge silencer. The PPF for these operating conditions is 252 Hz. There is a strong pulsation peak at 1506 Hz, which is six times the fundamental PPF. The magnitude of the pulsation at high order multiples of the fundamental PPF indicates that these frequencies may not be pure harmonics and are the result of a resonance at the compressor discharge. The acoustic system at

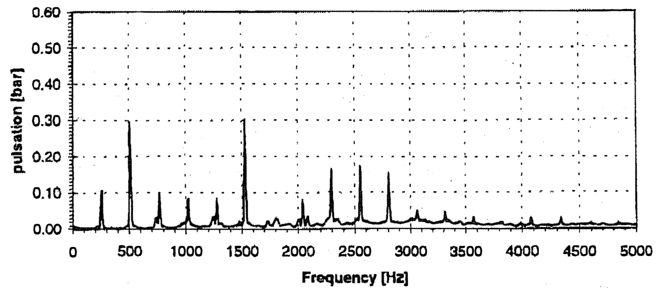

Figure 2. Discharge Pressure Pulsation Frequency Spectrum.

the compressor discharge includes the plenum from the rotor end-wall to the discharge flange and the inlet chamber of the silencer. Evaluation of the acoustic resonance of the discharge piping system should include analysis of the discharge plenum coupled to the PD.

Many piping sections or pressure vessels have devices attached to the wall that can act as a side branch resonator. Side branch resonance can occur where there is a discrete volume that is open to the flow path of the gas. The resonant frequency at which a side branch responds is dependent on the length of the open volume. If the side branch is excited by a frequency that is an odd multiple of the fundamental resonance of the channel, then there is an acoustic node at the opening of the side branch. Side branch resonators can be used as attenuation devices when tuned to odd multiples of an excitation. If the excitation occurs at an even multiple, then an antinode is present. An antinode that is present at the opening of the side branch results in amplification of the excitation frequency. Instrument connections, inspection openings, and drains can all act as sources of resonance.

Influence of Process Conditions

The sound power at the discharge port can reach levels in excess of 150 dB under steady-state operating conditions. Changes in the process conditions can have a significant impact on the pulsation pressure level produced at the discharge port. The pressure inside the screw threads is fixed by the port geometry at the discharge end-wall of the compressor. If the downstream pressure in the process is set at a point that does not match that of the builtin volume ratio, then there is a pressure mismatch at the port opening resulting in an elevated pressure pulsation magnitude.

The fundamental pocket passing frequency changes as a function of compressor speed. Attenuation characteristics of the pulsation damper components are degraded when the compressor is operated off the optimum design speed. Design length of acoustic dampers such as choke tubes and side branches is dependent upon the wavelength of the frequencies of interest. The pulsation damper design can be modified to accommodate off-design conditions, such as speed variation, by changing the end condition of the tubes by tapering or perforations, but such modifications reduce the attenuation at the center frequency.

The local speed of sound in the process gas is dependent upon the molecular weight, temperature, and pressure.

$$c = \sqrt{\gamma (R/mw) T_k} \quad (1)$$

where:

c = Speed of sound m/s
γ = Ratio of specific heats
R = Universal gas constant
mw = Molecular weight
T_k = Temperature °Kelvin

If the process molecular weight varies from the design point in a short cycle, then the change in the attenuation effectiveness of the

PD is temporary and these conditions will not significantly alter the design of the PD. If the molecular weight varies continuously, then the resulting fluctuation in wavelength may require a modification to the PD design. The temperature of the gas will change the attenuation characteristics of a passive acoustic element. The transmission coefficient of a side branch in a pipe decreases as the temperature of the process gas increases. The dashed line in Figure 3 indicates the transmission coefficient of a side branch where the temperature is increased by 100°C. Processes that have fluctuations in the gas composition by design should be evaluated by the compressor and PD manufacturers for resonance and attenuation analysis.

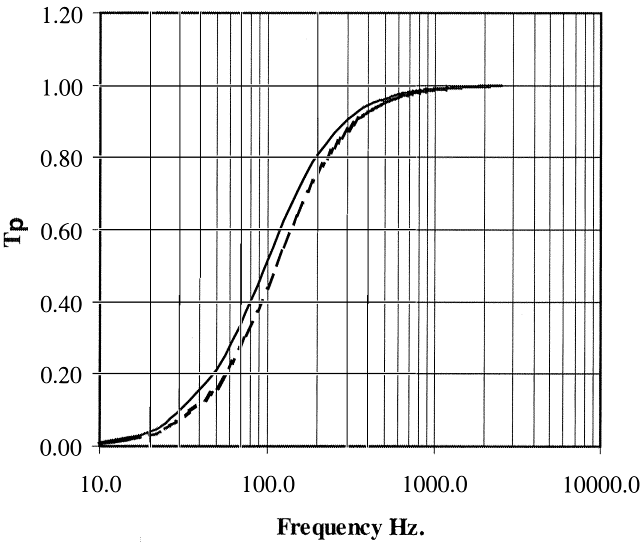

Figure 3. Temperature Dependence Transmission Coefficient Comparison in a Side Branch.

PULSATION DAMPER COMPONENT CONSTRUCTION

Pulsation dampers that are applied in process gas service are constructed from a combination of passive acoustic elements. Passive acoustic elements are those that have a fixed acoustic resonance characteristic. Examples of passive acoustic elements include open choke tubes, open and closed perforated tubes, orifice plates, baffles, and acoustic packing material, as shown in Figure 4. The tubes, orifice plates, and baffles have discrete acoustic cutoff frequencies that can be tuned for the PPF. Reduction of the pulsation amplitude for frequencies below 1000 Hz is not difficult to achieve using passive pulsation damper designs. The acoustic packing materials are broad band absorptive devices that are especially effective at high frequency attenuation. Absorptive packing materials such as fiberglass or glass beads must be contained in a mesh or perforated material in order to be effective. The perforated retention material may fail in fatigue and will generally not meet the three year service life expected in API 619 equipment. If the perforated material could be designed with a suitable fatigue life, then the packing material may settle, break apart, or become blocked due to solids carryover in the process. The packed materials are not considered as acceptable design components for most API applications due to these design limitations. The pulsation dampers used in API refinery process equipment are designed with passive acoustic elements, which are tuned to attenuate discrete frequencies.

Side Branch Resonance in Secondary Connections

Pulsation dampers are constructed with a number of secondary connections for pressure and temperature measurement as well as inspection ports. The process connection flanges and the adjoining tube section are considered as critical components in the acoustic

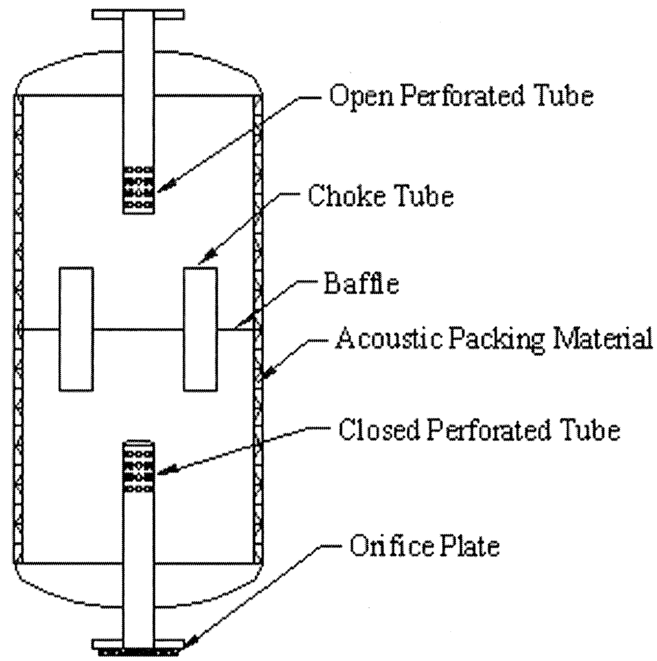

Figure 4. Passive Acoustic Element Designs.

design of the pulsation damper; however, the mechanical or acoustic influence of secondary connections is not always thoroughly evaluated.

Any opening in the wall of the pulsation damper or compressor plenum is a potential source of acoustic resonance. This resonance occurs at a discrete frequency that is a function of the opening geometry and the process conditions in the chamber. If the opening in the wall is closed at one end, then a standing wave is set up with an antinode at the closed end-wall. A tube that is closed at one end and designed to a length that is an odd multiple of the 1/4 wavelength of interest will have a node at the opening along the silencer wall. Tubes that are designed with nodes at the wall opening will dissipate some of the energy in the acoustic wave at the tuned frequency. If the tube length is coincident with an even multiple of the 1/4 wavelength, then an antinode is present at the wall and the tube will act as an amplifier at a multiple of the PPF. The closed end tube is created where there is a connection for pressure or temperature measurement. If a thermal well is extended into the process stream, a resonant cavity may still exist in the annulus created by the temperature probe and the piping connection through which it is inserted.

Pipe connections that are designed for a specific wavelength will operate well under conditions that produce pulsation at a multiple of the PPF. If the process conditions change the side branch acoustic resonant frequency, then the tube opening at the vessel side-wall may become excited.

Mechanical Integrity of Attachments to the Shell

The mechanical resonant frequency of each pulsation damper component should be considered as they are coupled to the shell at the attachment point. Components attached to the shell act as constraints and produce stress risers in the shell wall. If there is a shell vibration mode that is coincident with the PPF or any multiple, up to approximately 4000 Hz, then the fatigue life of the shell at these stress risers can be reduced significantly. Most tubes that connect the chambers in a PD are long enough to require some restraint to prevent large cantilever deflections at the baffle. These tubes are usually braced to the outer vessel shell wall by welded plate, as shown in Figure 5. The PD shown in Figures 5 and 6 was used in a service with fluctuation of molecular weight from 50 percent to 100 percent of the rated design value. A finite element analysis (FEA) of the shell identified natural frequencies of the shell, which were resonant at several multiples of the PPF. Ring testing of the vessel was conducted by placement of accelerometers along the shell wall and exciting the wall with an impact hammer at several places along the shell. The weld attachment is a point of high stress concentration along the shell and tube walls. The tube supports failed at the weld joint in six of the eight plates used for bracing. Investigation of the fracture surface revealed that the cracks were initiated at the surface of the weld and propagated into the parent metal of the shell and choke tubes. The tubes shown failed in less than 700 hours of service. Side branch connections are also stress risers that can result in failure of the pressure containment vessel. The PD shown in Figure 6 had a crack around a drain connection. Liquid seepage around this weld was one of the first indications of failure of the PD. Instrument connections usually consist of a weldalet attached to the vessel wall with a threaded pipe and instrument valve for isolation. Fatigue failure of these connections can occur in less than 1000 hours of operation and frequently results in the connection breaking off the vessel. The pressure vessel should post weld heat treatment attachment of any device including nameplates and instrument connections in order to reduce residual stresses in the heat affected zone. Design changes can be made to reduce the stress concentration at component attachment points. Modal analysis of the PD assembly is critical in determining the best attachment points. The shell mode shapes of the PD unit can be complex and ring testing is a very good method of verifying the modal analysis model. If process conditions change to the point that an acoustic resonance is set up in an opening to the shell, then a mechanical resonance at the same frequency can be very destructive.

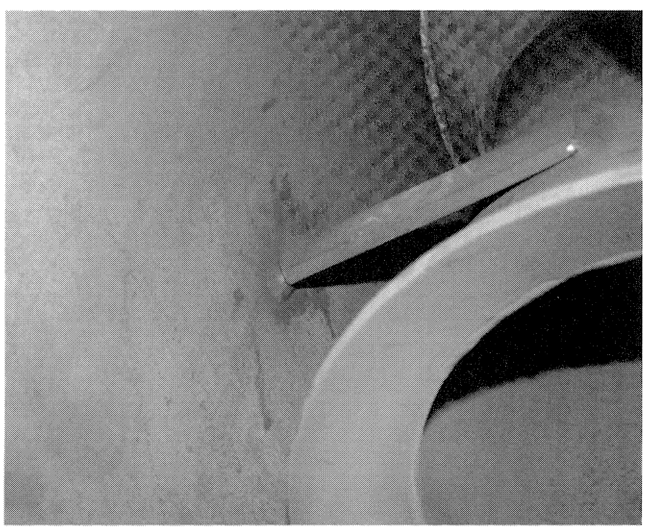

Figure 5. Choke Tube Supports.

COMPRESSOR PERFORMANCE

An increase in discharge pressure has a negligible impact on the inlet flow to the compressor. The compressor performance is measured by the inlet volume flow rate and power consumption. Predicted performance for a screw compressor, with 20 inch diameter rotors, indicates that an increase in the pressure at the discharge flange will increase the power requirement and decrease the flow, as shown in Figure 7. A 1 percent increase in the discharge pressure results in an increase in power of 1 percent for the case shown. The reduction in flow, for a similar pressure rise, is less than 0.1 percent. The pressure drop at the inlet flange to the compressor has a greater impact on the inlet volume flow rate than the pressure at the discharge. A plot showing the influence of inlet pressure on the flow rate is shown in Figure 8. A limit of 1 percent

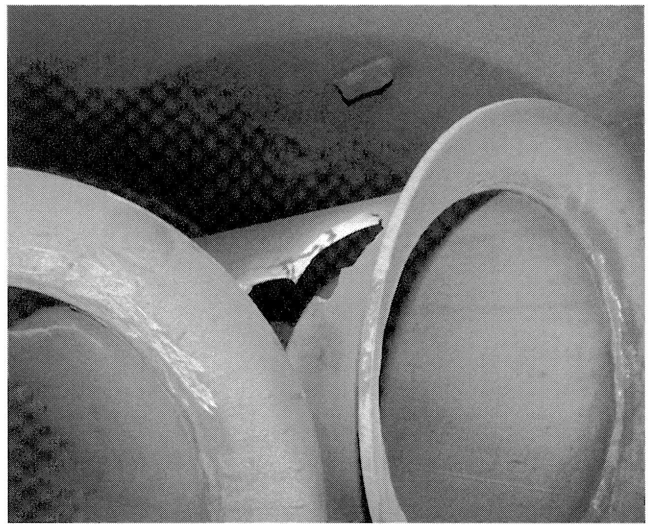

Figure 6. Choke Tube Support Failure.

pressure drop across the PD (as recommended by API) is intended to minimize the impact that the PD has on the compressor performance. The 1 percent limit at the inlet is a good specification to follow, and in practice the actual pressure drop across the inlet PD is usually less than 1 percent. The attenuation cutoff in many pulsation damper designs is limited to approximately 1000 Hz, in order to meet the 1 percent pressure drop specification. Passive attenuation of higher frequencies requires silencer components with reduced open areas resulting in higher pressure drop. High frequency pulsation at the discharge of the compressor requires that the design attenuation cutoff frequency should be greater than 3500 Hz. The inlet pressure pulsation amplitude that is seen upstream of the PD is not high enough to be a concern. Increasing the discharge pressure drop will increase the power requirements and operating costs, but these drawbacks are usually offset by reduced downtime and repair costs associated with piping failures.

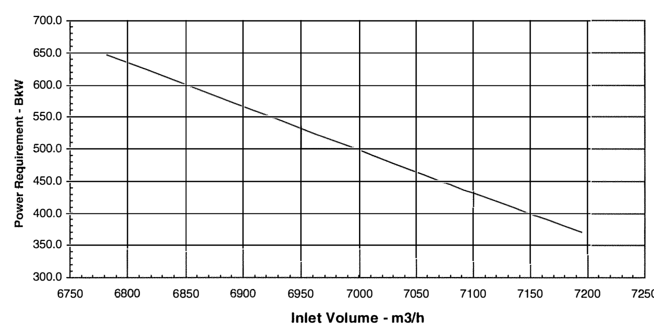

Figure 7. Compressor Flow Versus Discharge Pressure.

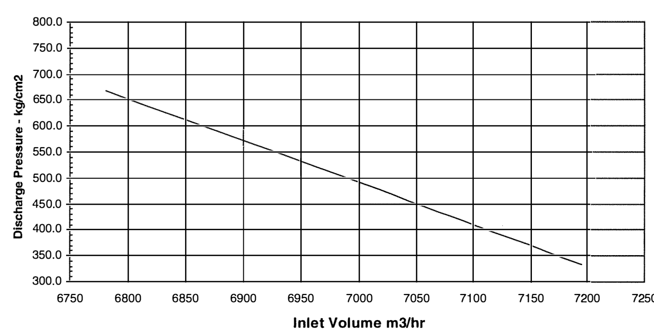

Figure 8. Compressor Flow Versus Inlet Pressure.

DOWNSTREAM PROCESS EQUIPMENT DESIGN CONSIDERATIONS

The principal purpose of API design requirements regarding pulsation reduction is to protect the piping downstream of the compressor discharge. Failure of components in piping systems does not always result in fracture or rupture of the piping wall. Electronic instrumentation, sight gauges, and loose bolts or fittings can be caused by excessive piping pulsation. High frequency pulsation may be strong enough to drive piping attachments into resonance. Piping attachments such as thermal wells, pressure taps, and brackets can easily be excited by an acoustic pulsation. The piping or PD shell can have a resonance at a multiple of the pocketing frequency and the dynamic displacement of the shell can be excessive when driven by a coincident acoustic frequency. The piping attachment weldament is a point of high stress concentration.

Most instrumentation connections can be isolated from the process and replaced with no down time; however, the stub-out connection at the piping wall is the most susceptible point of failure. Proper support of instrument line connections reduces the mass supported at the pipe connection and the displacement of the instrument line. Threaded connections have high stress concentrations at the root of the thread. Reducing the number of threaded connections will substantially increase the fatigue life of instrument lines.

Pipe supports are another point of stress concentration in the process piping. A ringing mode in the pipe may result in high stresses at an attachment point. Many structural welded connections are not stress relieved and reinforcement or redesign of these connections may be necessary.

Piping failures have occurred in processes downstream of a pulsation damper, even though the damper was designed to the proper industry standard. Pulsation levels that exist in OFS applications can produce pulsating pressure levels capable of exciting high order piping modes. Pulsation dampers designed in accordance to API 619 may not be adequate in reducing the pulsation (at frequencies greater than 1000 Hz) to an acceptable level. Changes in the PD design should be the first step in improvement of piping reliability, but careful analysis and design of attachment details will provide a much improved system design.

SOLUTIONS

Reduction of pressure pulsation in any process should start at the source. There are several methods that are in use for reducing the pressure pulsation level at the compressor discharge. One method involves bypass of some of the compressed gas within the rotor threads, to the low-pressure side of the thread just before opening at the discharge port. This method of discharge bypass is not very effective and results in a loss in volumetric efficiency. Reducing the pulsation level exiting the rotor threads lowers the pulsating pressure levels seen at the PD. Another method involves passive feedback of discharge pulsation to the inlet plenum out of phase with the inlet pulsation. The inlet pulsation is of much lower magnitude, making the phase cancellation marginally effective. Optimization of the discharge port to release the gas from the threads into the plenum is the most effective means of reducing pressure pulsation at the discharge. Port optimization also has the added benefit of improved efficiency.

Communication of the process operating conditions to the equipment manufacturers is essential in the procurement process. Point conditions are useful for determining steady-state performance, but off- design conditions can have a substantial impact on the compressor and PD performance. A description of process operating and startup procedures will go a long way toward a thorough design audit of the conditions that the equipment will see.

Simple design changes in the PD or process piping can result in significant improvement in fatigue life. Weldalets that are used for instrument connections should be gusseted or replaced with nipolets. The nipolet has a male connection, which can be connected to the instrument valve and does not require an intermediate threaded pipe. The reduction in length reduces the moment applied by the instrument valve to the welded connection. Stress relief of all process piping connections between the compressor and the PD should be mandatory.

Thorough mechanical analysis of the PD and the downstream piping is necessary in order to avoid mechanical resonance over the full range of operating conditions. Acoustic analysis of the downstream piping system is another improvement in the system design that will identify possible sources of acoustic resonance.

The API 619 specification for silencers and pulsation dampers requires that the design attenuate frequencies in the audible hearing range without exceeding the pressure drop limit. An increase in the pressure drop limit to 2 percent to 3 percent can increase the cutoff frequency to approximately 2000 Hz without an appreciable loss in compressor performance.

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